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DESCRIPTION

ROTARY FLUID MACHINE

FIELD OF THE INVENTION

The present invention relates to a rotary fluid machine in which opposite ends of a rotor are rotatably supported in a casing via a first bearing and a second bearing, and energy conversion means for interconverting pressure energy of a working medium and mechanical energy of the rotating rotor is provided in the rotor.

BACKGROUND ART

Such a rotary fluid machine is known from Japanese Patent Application Laid-open No. 2002-256805. This rotary fluid machine converts the pressure energy of high temperature, high pressure steam into mechanical energy for rotating an output shaft via two axial piston cylinder groups arranged in radially inner and outer stages, and axially opposite ends of a rotor thereof are each rotatably supported in a casing via one angular bearing.

The pair of angular bearings supporting in the casing axially opposite ends of the rotor of the above-mentioned conventional rotary fluid machine not only support a radial load of the rotor but also support an axial load so as to position the rotor in the axial direction. This results in the problems that, due to a difference in the coefficient of thermal expansion between the rotor and the casing, the bearing gap between the pair of angular bearings changes thus degrading the durability, the support of the rotor becomes unstable thus inhibiting smooth rotation, and the dead volume of the axial piston cylinder groups (a space between the top of a piston at top dead center and the top of a cylinder) varies thus changing the volume ratio (expansion ratio). In order to solve these problems, consideration has been given to using, among the pair of bearings supporting axially opposite ends of the rotor in the casing, only one of the bearings for supporting the axial load of the rotor, thus absorbing the difference in coefficient of thermal expansion between the rotor and the casing.

However, even by supporting the axial load of the rotor with only one of the bearings as described above, since the bearing is generally formed from an iron-based material having a small coefficient of thermal expansion from the viewpoint of strength and rigidity, and the casing is formed from an aluminum-based material having a large coefficient of thermal expansion from the viewpoint of light weight, etc., as shown in FIG. 20 an axial gap β is generated between the casing and the bearing when the rotary fluid machine is hot, this gap β results in an axial displacement of the rotor relative to the casing, and there is thus a possibility that the sealability of a rotary valve for supplying and discharging the working medium to and from the rotor might be degraded.

In order to prevent the axial gap from being generated between the casing and the bearing when the rotary fluid machine is hot, the bearing may be assembled to the casing in a state in which an axially compressive load is applied to the bearing, but this gives rise to the problem that the frictional resistance of the bearing to which the compressive load is applied might increase.

DISCLOSURE OF INVENTION

The present invention has been achieved under the above-mentioned circumstances, and it is an object thereof to solve the above-mentioned problems caused by a difference in the amount of thermal expansion between a casing and a rotor of a rotary fluid machine.

In order to attain this object, in accordance with a first aspect of the present invention, there is provided a rotary fluid machine in which opposite ends of a rotor are rotatably supported in a casing via a first bearing and a second bearing, and energy conversion means for interconverting pressure energy of a working medium and mechanical energy of the rotating rotor is provided in the rotor, characterized in that among the first bearing and the second bearing, the axial load can be supported by only the first bearing.

In accordance with this arrangement, since the axial load can be supported by only the first bearing among the first bearing and the second bearing, which rotatably

support opposite ends of the rotor in the casing, it is possible to prevent an axial load from being applied between the second bearing and the rotor due to a difference in the amount of axial thermal expansion between the casing and the rotor, while enabling the rotor to be axially positioned relative to the casing by only the first bearing. Because of this, not only is it possible to prevent the durability from being degraded by a preload on the first and second bearings being decreased by a difference in the amount of axial thermal expansion between the casing and the rotor or by a fluctuation in the load at high temperature, and particularly at low temperature, accompanying a change in the gap between the bearings, but it is also possible to ensure smooth rotation by stabilizing the support of the rotor by the first and second bearings and, moreover, by reducing the variation in dead volume of the energy conversion means it is possible to ensure that there is a desired volume ratio (expansion ratio or compression ratio).

In accordance with a second aspect of the present invention, in addition to the first aspect, the rotary fluid machine is an expander, and the energy conversion means is an axial piston cylinder group.

In accordance with this arrangement, since the energy conversion means of the expander for converting the pressure energy into mechanical energy is formed from the axial piston cylinder group, which is large in the axial direction, even when the difference in the amount of axial thermal expansion between the casing and the rotor greatly increases due to the large difference in temperature between when the temperature is cold and when the temperature is hot, it is possible to prevent an excessively varying load from being applied to the first and second bearings. It is also possible to stabilize the dead volume between a piston and a cylinder and thus prevent the volume ratio (expansion ratio) of the expander from changing.

In accordance with a third aspect of the present invention, in addition to the first aspect, the rotary fluid machine is provided with a rotary valve for supplying and discharging the working medium to and from the rotor, the coefficient of thermal expansion of the rotor is set so as to be substantially the same as the coefficient of

thermal expansion of the first bearing, the coefficient of thermal expansion of the casing is set so as to be larger than the coefficient of thermal expansion of the rotor and the coefficient of thermal expansion of the first bearing, the first bearing is supported in the casing via a bearing holder, and the coefficient of thermal expansion of the bearing holder is set so as to be substantially the same as the coefficient of thermal expansion of the rotor and the coefficient of thermal expansion of the first bearing.

In accordance with this arrangement, since the coefficient of thermal expansion of the rotor and the coefficient of thermal expansion of the first bearing are made substantially the same, the coefficient of thermal expansion of the casing is made larger than the coefficient of thermal expansion of the rotor and the coefficient of thermal expansion of the first bearing, the first bearing is supported in the casing via the bearing holder, and the coefficient of thermal expansion of the bearing holder is made substantially the same as the coefficient of thermal expansion of the rotor and the coefficient of thermal expansion of the first bearing, even when there is a difference in coefficient of thermal expansion between the casing and the first bearing, not only is it possible to prevent a gap from being generated between the first bearing and the bearing holder and prevent the sealability of the rotary valve from being degraded by the rotor moving axially due to the gap, but it is also possible to reduce the weight while ensuring a desired strength and rigidity.

In accordance with a fourth aspect of the present invention, in addition to the third aspect, the rotary fluid machine is an expander, and the energy conversion means is an axial piston cylinder group operated by a swash plate.

In accordance with this arrangement, since the energy conversion means of the expander for converting the pressure energy into mechanical energy is formed from the axial piston cylinder group, which is large in the axial direction, even when the difference in the amount of axial thermal expansion between the casing and the rotor greatly increases due to the large difference in temperature between when the

temperature is cold and when the temperature is hot, it is possible to prevent an excessively varying load from being applied to the first and second bearings. It is also possible to stabilize the dead volume between a piston and a cylinder and thus prevent the volume ratio (expansion ratio) of the expander from changing.

In accordance with a fifth aspect of the present invention, in addition to the fourth aspect, the swash plate is supported in the casing via a swash plate holder, and the coefficient of thermal expansion of the swash plate holder is set so as to be substantially the same as the coefficient of thermal expansion of the bearing holder.

In accordance with this arrangement, since the coefficient of thermal expansion of the swash plate holder for supporting the swash plate in the casing is made substantially the same as the coefficient of thermal expansion of the bearing holder, it is possible to prevent displacement of the position where a piston of the axial piston cylinder group comes into contact with the swash plate, thus preventing seizure occurring or any increase in the frictional resistance and, moreover, it is possible to stabilize the positional relationship between the piston, which abuts against the swash plate, and the cylinder, which is provided in the rotor, and thus prevent yet more effectively the volume ratio (expansion ratio) of the expander from changing.

In accordance with a sixth aspect of the present invention, in addition to the fifth aspect, the swash plate holder and the bearing holder are formed from the same member.

In accordance with this arrangement, since the swash plate holder and the bearing holder are formed from the same member, not only is it possible to prevent yet more effectively the volume ratio (expansion ratio) of the expander from changing, but it is also possible to reduce the number of components compared with a case in which they are formed from separate members.

Combined angular bearings 23f and 23r of an embodiment correspond to the first bearing of the present invention, and a radial bearing 24 of the embodiment corresponds to the second bearing of the present invention.

BRIEF DESCRIPTION OF DRAWINGS

FIG. 1 to FIG. 13 show a first embodiment of the present invention; FIG. 1 is a vertical sectional view of an expander, FIG. 2 is a sectional view along line 2-2 in FIG. 1, FIG. 3 is a view from arrowed line 3-3 in FIG. 1, FIG. 4 is an enlarged view of part 4 in FIG. 1, FIG. 5 is an enlarged view of part 5 in FIG. 1, FIG. 6 is an exploded perspective view of a rotor, FIG. 7 is a sectional view along line 7-7 in FIG. 4, FIG. 8 is a sectional view along line 8-8 in FIG. 4, FIG. 9 is an enlarged view of part 9 in FIG. 4, FIG. 10 is a sectional view along line 10-10 in FIG. 5, FIG. 11 is a sectional view along line 11-11 in FIG. 5, FIG. 12 is a sectional view along line 12-12 in FIG. 5, and FIG. 13 is a sectional view along line 13-13 in FIG. 5.

FIG. 14 and FIG. 15 show a second embodiment of the present invention; FIG. 13 is a view corresponding to FIG. 1, and FIG. 15 is a graph showing the relationship between increase in temperature and size of a gap of combined angular bearings.

FIG. 16 to FIG. 19 show a third embodiment of the present invention; FIG. 16 is an enlarged view of the surroundings of combined angular bearings of an expander, FIG. 17 is a diagram for explaining the reason for the volume ratio of the expander changing due to thermal expansion, FIG. 18 is a graph for comparing the temperature of a zone C1 and that of a zone C2 of the expander, and FIG. 19 is a graph showing changes in the dead volume of the axial piston cylinder group with respect to the temperature of the zone C2.

FIG. 20 is a diagram for explaining a gap generated between a casing and a bearing.

BEST MODE FOR CARRYING OUT THE INVENTION

A first embodiment of the present invention is explained below by reference to the attached drawings.

As shown in FIG. 1 to FIG. 9, an expander E of this embodiment is used in, for example, a Rankine cycle system, and converts the thermal energy and the pressure energy of high-temperature, high-pressure steam as a working medium into

mechanical energy and outputs it. A casing 11 of the expander E is formed from a casing main body 12, a front cover 15 joined via a seal 13 to a front opening of the casing main body 12 by a plurality of bolts 14, a rear cover 18 joined via a seal 16 to a rear opening of the casing main body 12 by a plurality of bolts 17, and an oil pan 21 joined via a seal 19 to a lower opening of the casing main body 12 by a plurality of bolts 20.

A rotor 22 arranged rotatably around an axis L extending in the fore-and-aft direction in the center of the casing 11 has a front part thereof supported by combined angular bearings 23f and 23r provided in the front cover 15 and a rear part thereof supported by a radial bearing 24 provided in the casing main body 12. A swash plate holder 28 is formed integrally with a rear face of the front cover 15, and a swash plate 31 is rotatably supported by the swash plate holder 28 via an angular bearing 30. The axis of the swash plate 31 is inclined relative to the axis L of the rotor 22, and the angle of inclination is fixed.

The rotor 22 includes an output shaft 32 supported in the front cover 15 by the combined angular bearings 23f and 23r, three sleeve support flanges 33, 34, and 35 formed integrally with a rear part of the output shaft 32 via cutouts 57 and 58 (see FIG. 4 and FIG. 9) having predetermined widths, a rotor head 38 that is joined by a plurality of bolts 37 to the rear sleeve support flange 35 via a metal gasket 36 and is supported in the casing main body 12 by the radial bearing 24, and a heat-insulating cover 40 that is fitted over the three sleeve support flanges 33, 34, and 35 from the front and joined to the front sleeve support flange 33 by a plurality of bolts 39. Sets of five sleeve support holes 33a, 34a, and 35a are formed in the three sleeve support flanges 33, 34, and 35 respectively at intervals of 72° around the axis L, and five cylinder sleeves 41 are fitted into the sleeve support holes 33a, 34a, and 35a from the rear. A flange 41a is formed on the rear end of each of the cylinder sleeves 41, and axial positioning is carried out by abutting this flange 41a against the metal gasket 36 while fitting the flange 41a into a step 35b formed in the sleeve support holes 35a of the rear sleeve support flange 35 (see FIG. 9). A piston 42 is slidably

fitted within each of the cylinder sleeves 41, the front end of the piston 42 abutting against a dimple 31a formed on the swash plate 31, and a steam expansion chamber 43 is defined between the rear end of the piston 42 and the rotor head 38.

A plate-shaped bearing holder 92 is superimposed on a front face of the front cover 15 via a seal 91 and fixed thereto by means of bolts 93, and a pump body 95 is superimposed on a front face of the bearing holder 92 via a seal 94 and fixed thereto by means of bolts 96. The combined angular bearings 23f and 23r are held between a step of the front cover 15 and the bearing holder 92, thereby fixing them in the axis L direction.

A shim 97 having a predetermined thickness is held between the inner race of the combined angular bearings 23f and 23r and a flange 32d formed on the output shaft 32 supporting the combined angular bearings 23f and 23r, and the inner race of the combined angular bearings 23f and 23r is tightened by a nut 98 screwed around the outer periphery of the output shaft 32. As a result, the output shaft 32 is positioned in the axis L direction relative to the combined angular bearings 23f and 23r, that is, relative to the casing 11.

The combined angular bearings 23f and 23r are mounted in mutually reversed directions, and not only support the output shaft 32 in the radial direction but also support it so that it does not move in the axis L direction. That is, one of the combined angular bearings 23f is disposed so as to restrain the output shaft 32 from moving forward, and the other of the combined angular bearings 23r is disposed so as to restrain the output shaft 32 from moving rearward.

Since the combined angular bearings 23f and 23r are used as bearings for supporting a front part of the rotor 22, one of the loads generated in opposite directions along the axis L in the expansion chambers 43 under predetermined operating conditions of the expander E is transferred to the inner race of the combined angular bearings 23f and 23r via the rotor 22, and the other thereof is transferred to the outer race of the combined angular bearings 23f and 23r via the swash plate 31 and the swash plate holder 28 of the front cover 15. These two loads

compress the swash plate holder 28 of the front cover 15, the swash plate holder 28 being held between the angular bearing 30 supporting the swash plate 31 and the combined angular bearings 23f and 23r supporting the rotor 22, and the rigidity of the mechanical part becomes high. Moreover, as in this embodiment, forming the swash plate holder 28 integrally with the front cover 15 enables the rigidity to be further enhanced and the structure to be made simple.

Furthermore, by incorporating the angular bearing 30 supporting the swash plate 31 and the combined angular bearings 23f and 23r supporting the rotor 22 into the front cover 15, the assembly operation can be carried out in the form of units such as 'rotor 22 and pistons 42', 'assembly of front cover 15', and 'pump body 95', and the efficiency of operations such as recombining the pistons 42 and exchanging the oil pump 49 is improved.

A radial bearing 24 supporting the rotor head 38, which forms a rear end part of the rotor 22, is a normal ball bearing supporting only a radial load, and a gap α (see FIG. 5) is formed between the rotor head 38 and the inner race of the radial bearing 24 so that the rotor head 38 can slide against the radial bearing 24 in the axis L direction.

An oil passage 32a is formed so as to extend along the axis L within the output shaft 32, which is integral with the rotor 22, and the front end of the oil passage 32a branches in a radial direction and communicates with an annular channel 32b on the outer periphery of the output shaft 32. An oil passage blocking member 45 is screwed into the inner periphery of the oil passage 32a via a seal 44 at a position that is radially inside the middle sleeve support flange 34 of the rotor 22, and a plurality of oil holes 32c extending radially outward from the oil passage 32a in the vicinity of the oil passage blocking member 45 open on the outer periphery of the output shaft 32.

A trochoidal oil pump 49 is disposed between a recess 95a provided in a front face of the pump body 95 and a pump cover 48 fixed via a seal 46 to the front face of the pump body 95 by a plurality of bolts 47, and includes an outer rotor 50 that is

rotatably fitted in the recess 95a, and an inner rotor 51 that is fixed to the outer periphery of the output shaft 32 and meshes with the outer rotor 50. An internal space of the oil pan 21 communicates with an intake port 53 of the oil pump 49 via an oil pipe 52 and an oil passage 95b of the pump body 95, and a discharge port 54 of the oil pump 49 communicates with the annular channel 32b of the output shaft 32 via an oil passage 95c of the pump body 95.

The piston 42, which is slidably fitted into the cylinder sleeve 41, is formed from an end part 61, a middle part 62, and a top part 63. The end part 61 is a member having a spherical part 61a that abuts against the dimple 31a of the swash plate 31, and is joined by welding to the forward end of the middle part 62. The middle part 62 is a cylindrical member having a large volume hollow space 62a; an outer peripheral part of the middle part 62 close to the top part 63 has a small diameter part 62b whose diameter is slightly reduced, a plurality of oil holes 62c are formed so as to run radially through the small diameter part 62b, and a plurality of spiral oil channels 62d are formed in an outer peripheral part that is present forward of the small diameter part 62b. The top part 63 faces the expansion chamber 43 and is formed integrally with the middle part 62, and a heat-insulating space 65 (see FIG. 9) is formed between a dividing wall 63a formed on an inner face of the top part 63 and a cover member 64 fitted into and welded to a rear end face of the top part 63. Two compression rings 66 and one oil ring 67 are mounted on the outer periphery of the top part 63, and an oil ring channel 63b into which the oil ring 67 is fitted communicates with the hollow space 62a of the middle part 62 via a plurality of oil holes 63c.

The end part 61 and the middle part 62 of the piston 42 are made of high-carbon steel, and the top part 63 is made of stainless steel; among these, the end part 61 is subjected to induction hardening, whereas the middle part 62 is subjected to hardening. As a result, high surface pressure resistance can be imparted to the end part 61, which abuts against the swash plate 31 at a high surface pressure, abrasion resistance can be imparted to the middle part 62, which is in sliding contact

with the cylinder sleeve 41 under severe lubrication conditions, and heat resistance and corrosion resistance can be imparted to the top part 63, which faces the expansion chamber 43 and is exposed to high temperature and high pressure.

An annular channel 41b is formed on the outer periphery of a middle part of the cylinder sleeve 41 (see FIG. 6 and FIG. 9), and a plurality of oil holes 41c are formed in the annular channel 41b. Regardless of where rotationally the cylinder sleeve 41 is mounted, the oil holes 32c formed in the output shaft 32 and oil holes 34b formed in the middle sleeve support flange 34 of the rotor 22 (see FIG. 4 and FIG. 6) communicate with the annular channel 41b. A space 68 formed between the heat-insulating cover 40 and the front and rear sleeve support flanges 33 and 35 of the rotor 22 communicates with the internal space of the casing 11 via oil holes 40a (see FIG. 4 and FIG. 7) formed in the heat-insulating cover 40.

An annular cover member 69 is welded to the front, or expansion chamber 43 side, of the rotor head 38, which is joined to the rear face of the front sleeve support flange 33 of the rotor 22 by the bolts 37, and an annular heat-insulating space 70 (see FIG. 9) is defined at the back, or rear face, of the cover member 69. The rotor head 38 is positioned rotationally relative to the rear sleeve support flange 35 by a knock pin 55.

The five cylinder sleeves 41 and the five pistons 42 form an axial piston cylinder group 56 of the present invention.

The structure of a rotary valve 71 for the supply and discharge of steam to and from the five expansion chambers 43 of the rotor 22 is now explained by reference to FIG. 5, and FIG. 10 to FIG. 13.

As shown in FIG. 5, the rotary valve 71, which is disposed along the axis L of the rotor 22, includes a valve main body 72, a stationary valve plate 73, and a movable valve plate 74. The movable valve plate 74 is fixed to a rear face of the rotor 22 by a bolt 76 screwed into the oil passage blocking member 45 (see FIG. 4) while being positioned in the rotational direction by a knock pin 75. The bolt 76 also has the function of fixing the rotor head 38 to the output shaft 32.

As is clear from FIG. 5, the stationary valve plate 73, which abuts against the movable valve plate 74 via flat sliding surfaces 77, is fixed to the center of a front face of the valve main body 72 by one bolt 78, and also to an outer peripheral part of the valve main body 72 by an annular fixing ring 79 and a plurality of bolts 80. During this process, a step 79a formed on the inner periphery of the fixing ring 79 is press-fitted in a spigot-joint manner around the outer periphery of the stationary valve plate 73, and a step 79b formed on the outer periphery of the fixing ring 79 is press-fitted in a spigot-joint manner around the outer periphery of the valve main body 72, thereby ensuring that the stationary valve plate 73 is coaxial with the valve main body 72. A knock pin 81 is disposed between the valve main body 72 and the stationary valve plate 73, and determines the position of the stationary valve plate 73 in the rotational direction.

When the rotor 22 rotates, the movable valve plate 74 and the stationary valve plate 73 therefore rotate relative to each other on the sliding surfaces 77 in a state in which they are in intimate contact with each other. The stationary valve plate 73 and the movable valve plate 74 are made of a material having excellent durability, such as carbon or a ceramic, and the durability can be further improved by providing or coating the sliding surfaces 77 with a member having heat resistance, lubricating properties, corrosion resistance, and abrasion resistance.

The valve main body 72, which is made of stainless steel, is a stepped cylindrical member having a large diameter part 72a and a small diameter part 72b; outer peripheral faces of the large diameter part 72a and the small diameter part 72b are fitted slidably in the axial L direction into circular cross-section support faces 18a and 18b of the rear cover 18 via seals 82 and 83 respectively, and positioned in the rotational direction by fitting a pin 84 implanted in an outer peripheral face of the valve main body 72 into a cutout 18c formed in the axial L direction in the rear cover 18. A plurality of preload springs 85 are supported in the rear cover 18 so as to surround the axis L, and the valve main body 72, which has a step 72c between the large diameter part 72a and the small diameter part 72b pushed by these preload

springs 85, is biased forward so as to put the sliding surfaces 77 of the stationary valve plate 73 and the movable valve plate 74 in intimate contact.

A steam supply pipe 86 connected to a rear face of the valve main body 72 communicates with the sliding surfaces 77 via a first steam passage P1 formed in the interior of the valve main body 72 and a second steam passage P2 formed in the stationary valve plate 73. A steam discharge chamber 88 sealed by a seal 87 is formed between the casing main body 12, the rear cover 18, and the rotor 22, and this steam discharge chamber 88 communicates with the sliding surfaces 77 via sixth and seventh steam passages P6 and P7 formed in the interior of the valve main body 72 and a fifth steam passage P5 formed in the stationary valve plate 73. Provided on surfaces where the valve main body 72 and the stationary valve plate 73 are joined are a seal 89 surrounding a part where the first and second steam passages P1 and P2 are connected to each other and a seal 90 surrounding a part where the fifth and sixth steam passages P5 and P6 are connected to each other.

Five third steam passages P3 disposed at equal intervals so as to surround the axis L run through the movable valve plate 74, and opposite ends of five fourth steam passages P4 formed in the rotor 22 so as to surround the axis L communicate with the third steam passages P3 and the expansion chambers 43. The part of the second steam passage P2 opening on the sliding surface 77 is circular, whereas the part of the fifth steam passage P5 opening on the sliding surface 77 has an arc shape with the axis L as its center.

The operation of the expander E of the embodiment having the above-mentioned arrangement is now explained.

High temperature, high pressure steam generated by heating water in an evaporator reaches the sliding surfaces 77 of the stationary valve plate 73 with the movable valve plate 74 from the steam supply pipe 86 via the first steam passage P1 formed in the valve main body 72 of the rotary valve 71 and the second steam passage P2 formed in the stationary valve plate 73, which is integral with the valve main body 72. The second steam passage P2 opening on the sliding surface 77

communicates momentarily during a predetermined intake period with the corresponding third steam passage P3 formed in the movable valve plate 74, which rotates integrally with the rotor 22, and the high temperature, high pressure steam is supplied, via the fourth steam passage P4 formed in the rotor 22, from the third steam passage P3 to the expansion chamber 43 within the cylinder sleeve 41. Even after the communication between the second steam passage P2 and the third steam passage P3 has been blocked due to rotation of the rotor 22, the high temperature, high pressure steam expands within the expansion chamber 43 and causes the piston 42 fitted in the cylinder sleeve 41 to be pushed forward from top dead center toward bottom dead center, and the end part 61 at the front end of the piston 42 pushes against the dimple 31a of the swash plate 31. As a result, a reaction force that the piston 42 receives from the swash plate 31 gives a rotational torque to the rotor 22. For each one fifth of a revolution of the rotor 22, the high temperature, high pressure steam is supplied into a fresh adjoining expansion chamber 43, thus continuously rotating the rotor 22.

While the piston 42, having reached bottom dead center accompanying rotation of the rotor 22, retreats toward top dead center by being pushed by the swash plate 31, the low temperature, low pressure steam pushed out of the expansion chamber 43 is discharged into the steam discharge chamber 88 via the fourth steam passage P4 of the rotor 22, the third steam passage P3 of the movable valve plate 74, the sliding surfaces 77, the arc-shaped fifth steam passage P5 of the stationary valve plate 73, and the sixth and seventh steam passages P6 and P7 of the valve main body 72, and is supplied therefrom into a condenser. The oil pump 49 provided on the output shaft 32 operates accompanying rotation of the rotor 22, and oil is taken in from the oil pan 21 via the oil pipe 52, the oil passage 95b of the pump body 95, and the intake port 53, discharged from the discharge port 54, and supplied to a space between the cylinder sleeve 41 and the small diameter part 62b formed in the middle part 62 of the piston 42 via the oil passage 95c of the pump body 95, the oil passage 32a of the output shaft 32, the annular channel 32b of the

output shaft 32, the oil holes 32c of the output shaft 32, the annular channel 41b of the cylinder sleeve 41, and the oil holes 41c of the cylinder sleeve 41. A portion of the oil retained by the small diameter part 62b flows into the spiral oil channels 62d formed in the middle part 62 of the piston 42 and lubricates the surface that slides against the cylinder sleeve 41, and another portion of the oil lubricates surfaces of the compression rings 66 and the oil ring 67 provided at the top part 63 of the piston 42 that slide against the cylinder sleeve 41.

Since water formed in the expansion chamber 43 by condensation of a portion of the supplied high temperature, high pressure steam inevitably enters between the sliding surfaces of the cylinder sleeve 41 and the piston 42 and contaminates the oil, the lubrication conditions of the sliding surfaces are severe, but by supplying a necessary amount of oil directly to the sliding surfaces of the cylinder sleeve 41 and the piston 42 from the oil pump 49 via the interior of the output shaft 32, it is possible to maintain a sufficient oil film, thereby ensuring the lubrication performance and enabling the dimensions of the oil pump 49 to be reduced.

Oil scraped off the surface of the cylinder sleeve 41 that the piston 42 slides against by the oil ring 67 flows from the oil holes 63c formed in the base of the oil ring channel 63b into the hollow space 62a within the piston 42. The hollow space 62a communicates with the interior of the cylinder sleeve 41 via the plurality of oil holes 62c running through the middle part 62 of the piston 42, and the interior of the cylinder sleeve 41 communicates with the annular channel 41b on the outer periphery of the cylinder sleeve 41 via the plurality of oil holes 41c. Although the surroundings of the annular channel 41b are covered by the middle sleeve support flange 34 of the rotor 22, since the oil hole 34b is formed in the sleeve support flange 34, the oil within the hollow space 62a of the piston 42 is urged radially outward due to centrifugal force, discharged to the space 68 within the heat-insulating cover 40 via the oil hole 34b of the sleeve support flange 34, and returned therefrom to the oil pan 21 via the oil holes 40a of the heat-insulating cover 40. During this process, since the oil hole 34b is positioned toward the axis L relative to the radially outer

edge of the sleeve support flange 34, the oil that is present radially outside the oil hole 34b is retained in the hollow space 62a of the piston 42 by centrifugal force.

In this way, the oil retained in the hollow space 62a within the piston 42 and the oil retained in the small diameter part 62b on the outer periphery of the piston 42 is supplied from the small diameter part 62b to the top part 63 side during an expansion stroke in which the volume of the expansion chamber 43 increases, and is supplied from the small diameter part 62b to the end part 61 side during a compression stroke in which the volume of the expansion chamber 43 decreases, and it is therefore possible to ensure reliable lubrication over the entire axial region of the piston 42. Furthermore, as a result of the oil flowing within the hollow space 62a of the piston 42, the heat of the top part 63, which is exposed to high temperature, high pressure steam, is transmitted to the end part 61, which has a low temperature, and it is thus possible to avoid the temperature of the piston 42 increasing locally.

When high temperature, high pressure steam is supplied from the fourth steam passage P4 to the expansion chamber 43, since the heat-insulating space 65 is formed between the middle part 62 and the top part 63 of the piston 42, which faces the expansion chamber 43, and the heat-insulating space 70 is formed in the rotor head 38, which faces the expansion chamber 43, it is possible to minimize the escape of heat from the expansion chamber 43 to the piston 42 and the rotor head 38, thereby contributing to an improvement in the performance of the expander E. Furthermore, since the large volume hollow space 62a is formed within the piston 42, not only is it possible to reduce the weight of the piston 42, but it is also possible to reduce the heat capacity of the piston 42, thereby enabling the escape of heat from the expansion chamber 43 to be suppressed yet more effectively.

Since the expansion chamber 43 is sealed by interposing the metal gasket 36 between the rear sleeve support flange 35 and the rotor head 38, in comparison with a case in which the expansion chamber 43 is sealed via a thick annular seal, unnecessary volume around the seal can be reduced, thus ensuring that the

expander E has a large volume ratio (expansion ratio) and thereby improving the thermal efficiency, which enables the output to be increased. Moreover, since the cylinder sleeve 41 is formed separately from the rotor 22, the material of the cylinder sleeve 41 can be selected without being restricted by the material of the rotor 22, while taking into consideration the thermal conductivity, heat resistance, strength, abrasion resistance, etc., and, moreover, it is possible to replace only a worn or damaged cylinder sleeve 41, which is economical.

Furthermore, since the outer peripheral face of the cylinder sleeve 41 is exposed through the two cutouts 57 and 58 formed circumferentially in the outer peripheral face of the rotor 22, not only is it possible to reduce the weight of the rotor 22, but it is also possible to reduce the heat capacity of the rotor 22, thereby improving the thermal efficiency and, moreover, the cutouts 57 and 58 function as a heat-insulating space, thus suppressing the escape of heat from the cylinder sleeve 41. Furthermore, since the outer peripheral part of the rotor 22 is covered by the heat-insulating cover 40, it is possible to suppress the escape of heat from the cylinder sleeve 41 yet more effectively.

Since the rotary valve 71 supplies and discharges steam to and from the axial piston cylinder group 56 via the flat sliding surfaces 77 between the stationary valve plate 73 and the movable valve plate 74, it is possible to prevent the leakage of steam effectively. This is because the flat sliding surfaces 77 can easily be machined with high precision, and control of the clearance is easy compared with cylindrical sliding surfaces. Moreover, since a surface pressure is generated on the sliding surfaces 77 of the stationary valve plate 73 and the movable valve plate 74 by applying a preset load to the valve main body 72 by means of the plurality of preload springs 85, it is possible to suppress the leakage of steam past the sliding surfaces 77 yet more effectively.

Furthermore, since the valve main body 72 of the rotary valve 71 is made of stainless steel, which has a large coefficient of thermal expansion, and the stationary valve plate 73 fixed to the valve main body 72 is made of carbon or a ceramic, which

has a small coefficient of thermal expansion, there is the possibility that the centering between the two might be displaced due to a difference in the coefficients of thermal expansion, but since the fixing ring 79 is fixed to the valve main body 72 by means of the plurality of bolts 80 in a state in which the step 79a on the inner periphery of the fixing ring 79 is press-fitted in a spigot-joint manner over the outer periphery of the stationary valve plate 73 and the step 79b on the outer periphery of the fixing ring 79 is press-fitted in a spigot-joint manner over the outer periphery of the valve main body 72, it is possible to carry out precise centering of the stationary valve plate 73 relative to the valve main body 72 by the aligning action of the press-fitting of the fixing ring 79 and prevent the timing of supply and discharge of steam from deviating, thereby preventing deterioration in the performance of the expander E. Moreover, it is possible to make the abutting surfaces of the stationary valve plate 73 and the valve main body 72 come into intimate and uniform contact by virtue of the securing force of the bolts 80, thereby suppressing the leakage of steam past the abutting surfaces.

Moreover, since the rotary valve 71 can be attached to and removed from the casing main body 12 merely by removing the rear cover 18 from the casing main body 12, the ease of maintenance operations such as repair, cleaning, and replacement can be greatly improved. Furthermore, although the rotary valve 71 through which the high temperature, high pressure steam passes reaches a high temperature, since the swash plate 31 and the output shaft 32, where lubrication by oil is required, are disposed on the opposite side of the rotor 22 to the rotary valve 71, degradation of the lubrication performance of the swash plate 31 and the output shaft 32 due to heating of the oil by the heat of the rotating valve 71, which reaches a high temperature, can be prevented. Moreover, the oil also exhibits the function of cooling the rotary valve 71, thus preventing overheating.

When the expander E is assembled, it is necessary to adjust the size of the dead volume between the base (that is, the cover member 69 supported on the rotor head 38) of the cylinder sleeve 41 and the top of the piston 42, that is, the volume of

the operating chamber 43 when the piston 42 is at top dead center. Thinning the shim 97 disposed between the flange 32d of the output shaft 32 and the inner race of the combined angular bearings 23f and 23r makes the output shaft 32 shift forward (to the right in FIG. 1), and the rotor head 38 also shifts forward, but since the piston 42 is restricted by the swash plate 31 and cannot shift forward, the dead volume decreases. On the other hand, increasing the thickness of the shim 97 makes the rotor head 38 shift rearward (to the left in FIG. 1) together with the output shaft 32, and the dead volume therefore increases. As a result, the dead volume can be freely adjusted by exchanging only the shim 97, the number of steps required for adjusting the dead volume can be decreased, and a large amount of time can be saved.

Furthermore, the dead volume can be adjusted simply by inserting a single shim 97 having a predetermined thickness between the flange 32d of the output shaft 32 and the combined angular bearings 23f and 23r and tightening via the one nut 98 the rotor 22 into which the pistons 42 are incorporated and the front cover 15 into which the angular bearing 30 supporting the swash plate 31 and the combined angular bearings 23f and 23r supporting the rotor 22 are incorporated, and it is therefore possible to carry out the adjustment easily compared with a conventional case in which the thickness of two of front and rear shims is adjusted individually. Moreover, when the dead volume is adjusted, since the rotor 22 into which the pistons 42 are incorporated may remain assembled to the casing main body 12, post-adjustment checking of the dead volume can be carried out while directly viewing a state in which the pistons 42 and the swash plate 31 are in contact with each other.

As hereinbefore described, when the position of the output shaft 32 relative to the combined angular bearings 23f and 23r is adjusted in the fore-and-aft direction by changing the thickness of the shim 97, although the position of the rotor head 38 at the rear end part of the rotor 22 also shifts in the fore-and-aft direction, since the rotor head 38 is freely slidable in the axis L direction relative to the inner race of the

radial bearing 24 provided between the casing main body 12 and the rotor head 38, there is no problem in adjusting the position of the output shaft 32.

When the piston 42 is urged by the pressure of the high temperature, high pressure steam supplied to the expansion chamber 43 in a direction in which the piston 42 is pushed out of the cylinder sleeve 41, the pressing force of the piston 42 pushes the outer race of the combined angular bearings 23f and 23r forward (to the right in FIG. 1) via the swash plate 31, the angular bearing 30, the swash plate holder 28, and the front cover 15, and the pressing force of the cylinder sleeve 41, which is in the reverse direction to the pressing force of the piston 42, pushes the inner race of the combined angular bearings 23f and 23r rearward (to the left in FIG. 1) via the rotor head 38 and the output shaft 32. That is, the loads generated by the high temperature, high pressure steam supplied to the expansion chamber 43 are cancelled out within the combined angular bearings 23f and 23r and are not transferred to the casing main body 12.

Whereas the rotor 22, which is formed from the output shaft 32, the three sleeve support flanges 33, 34, and 35, the rotor head 38, and the heat-insulating cover 40, is made of an iron-based material, which has relatively small coefficient of thermal expansion, the casing 11, which supports the rotor 22 via the combined angular bearings 23f and 23r and the radial bearing 24, is made of an aluminum-based material, which has relatively large coefficient of thermal expansion, and as a result a difference is generated in the amount of thermal expansion, in the axis L direction in particular, between when the temperature of the expander E is low and when it is high.

The casing 11, which has a larger coefficient of thermal expansion than that of the rotor 22, expands more than the rotor 22 when the temperature is high and the dimension of the casing 11 in the axis L direction relatively increases, whereas when the temperature is low the casing 11 shrinks more and the dimension thereof in the axis L direction relatively decreases. Since at this time the casing 11 and the rotor 22 are positioned in the axis L direction via the combined angular bearings 23f and

23r, the difference in the amount of thermal expansion between two is absorbed by the rotor head 38 sliding against the inner race of the radial bearing 24, thus preventing an excessive load in the axis L direction from being applied to the combined angular bearings 23f and 23r, the radial bearing 24, and the rotor 22. This enables not only the durability of the combined angular bearings 23f and 23r and the radial bearing 24 to be improved, but also the rotor 22 to be supported stably and rotated smoothly and, moreover, it is possible to prevent the dead volume between the top of the cylinder sleeve 41 and the top of the piston 42 from varying accompanying a change in the temperature.

This is because, if the opposite ends of the rotor 22 were restrained in the casing 11 so that the rotor 22 could not move in the axial direction, when the temperature is low the casing 11 would shrink in the axis L direction relative to the rotor 22, the piston 42 whose head abuts against the swash plate 31 supported by the swash plate holder 28, which is a part of the casing 11, would be pushed rearward, and the rotor head 38 supported in the casing 11 via the radial bearing 24 would be pushed forward, and as a result the piston 42 would be pushed into the cylinder sleeve 41, thus decreasing the dead volume. On the other hand, when the temperature is high, the casing 11 would elongate in the axis L direction relative to the rotor 22, the piston 42 would be pulled out of the interior of the cylinder sleeve 41, thus increasing the dead volume, and the initial volume of high temperature, high pressure steam under normal operating conditions after completion of warm up would increase, that is, the thermal efficiency would be degraded due to a decrease in the volume ratio (expansion ratio) of the expander E.

In this embodiment, however, since the rotor 22 is floatingly supported in the axis L direction relative to the casing 11, an increase in the gap between the combined angular bearings 23f and 23r and the radial bearing 24 and a decrease in the preload are prevented, and the dead volume is prevented from fluctuating accompanying a change in temperature. This prevents any fluctuation in the volume ratio (expansion ratio) of the expander E and ensures stable performance.

In particular, in the expander E employing high temperature, high pressure steam as the working medium, since there is a large difference in temperature between when the temperature is high and when the temperature is low, the above-mentioned effect can be exhibited effectively. Furthermore, the difference in temperature between when the temperature is high and when the temperature is low is large in the vicinity of the rotary valve 71, to which the high temperature, high pressure steam is supplied, but since the rotor head 38 can slide in the axis L direction against the radial bearing 24 disposed on the side close to the rotary valve 71, the difference in coefficient of thermal expansion between the casing 11 and the rotor 22 can be absorbed without any problem.

Among the stationary valve plate 73 and the movable valve plate 74 of the rotary valve 71, since the stationary valve plate 73 supported in the casing 11 is urged by an elastic force of the preload springs 85 toward the movable valve plate 74 supported on the rotor 22, even when the positional relationship in the axis L direction between the casing 11 and the rotor 22 changes accompanying a change in the temperature, there is no possibility of the sealability of the sliding surfaces 77 of the stationary valve plate 73 and the movable valve plate 74 being impaired. Instead, an excess load is prevented from being applied to the combined angular bearings 23f and 23r and the radial bearing 24, the rotational surface of the rotor 22 is stabilized, the sealability of the sliding surfaces 77 is improved, and the amount of steam leakage can be reduced.

A second embodiment of the present invention is now explained by reference to FIG. 14 and FIG. 15. In the second embodiment, members corresponding to the above-mentioned members of the first embodiment are denoted by the same reference numerals and symbols as those in the first embodiment, and duplication of the explanation is omitted.

In the first embodiment, the combined angular bearings 23f and 23r are supported directly in the casing 11, but in the second embodiment combined angular bearings 23f and 23r are supported in a casing 11 via a bearing holder 99. That is, a

substantially cylindrical bearing holder 99 fitted into the inner periphery of a front cover 15 is fixed, together with a plate-shaped set plate 92 superimposed on a front face of the bearing holder 99, by bolts 93, and a pump body 95 is further superimposed on a front face of the front cover 15 via a seal 94 and fixed by bolts 96. The combined angular bearings 23f and 23r are therefore fixed in the axis L direction while being held between a step of the bearing holder 99 and the set plate 92.

The bearing holder 99, the set plate 92, and the combined angular bearings 23f and 23r are formed, as for a rotor 22, from an iron-based material having a relatively small coefficient of thermal expansion.

In accordance with this second embodiment, the combined angular bearings 23f and 23r, which are formed from an iron-based material having a relatively small coefficient of thermal expansion, are not supported directly in the casing 11, which is formed from an aluminum-based material having a relatively large coefficient of thermal expansion, but instead the combined angular bearings 23f and 23r are supported in the casing 11 via the bearing holder 99, which is made of an iron-based material and fixed to the casing 11, and even if there is a difference between the coefficient of thermal expansion of the casing 11 and the coefficient of thermal expansion of the combined angular bearings 23f and 23r, as shown in FIG. 15, the occurrence of a gap β (see FIG. 20) due to a difference in thermal elongation between the bearing holder 99 and the combined angular bearings 23f and 23r can be suppressed, and it is possible to prevent the rotor 22 from moving in the axis L direction as a result of this gap β and prevent the sealability of the sliding surfaces 77 of the rotary valve from deteriorating.

A third embodiment of the present invention is now explained by reference to FIG. 16 to FIG. 19. In the third embodiment, members corresponding to the above-mentioned members of the first and second embodiments are denoted by the same reference numerals and symbols as those in the first and second embodiments, and duplication of the explanation is omitted.

In the second embodiment, the swash plate holder 28 is formed integrally with the front cover 15, but in the third embodiment shown in FIG. 16, a swash plate holder 28 is separate from a front cover 15 and is formed integrally with a bearing holder 99. The integrated bearing holder 99 and swash plate holder 28, together with a set plate 92 fixed thereto by bolts 93, are fixed to the front cover 15 by bolts 100. The swash plate holder 28 and the bearing holder 99 are formed from an iron-based material having a small coefficient of thermal expansion, as for the bearing holder 99 of the second embodiment.

In accordance with this third embodiment, since the coefficient of thermal expansion of the swash plate holder 28 is smaller than the coefficient of thermal expansion of the front cover 15, which is formed from an aluminum-based material, displacement of the swash plate holder 28 relative to a casing 11 due to thermal elongation can be minimized, and displacement of the position where an end part 61 of a piston 42 comes into contact with a dimple 31a of a swash plate 31 can be prevented, thus preventing seizure occurring or any increase in the frictional resistance. Moreover, the positional relationship in the axis L direction between the piston 42 abutting against the swash plate 31 and a cylinder sleeve 41 provided on a rotor 22 can be stabilized, and the volume ratio (expansion ratio) of an expander E can be prevented yet more effectively from changing.

The reason therefor is explained below by reference to FIG. 17.

The left end of the combined angular bearings 23f and 23r is defined as a starting point for thermal elongation, and a section from this point to the top of the cylinder sleeve 41 of the rotor 22 is defined as a zone A1. The zone A1 is thus formed from a zone B1 corresponding to the rotor 22 and a zone C1 corresponding to an output shaft 32. A section from the starting point for thermal elongation to the top of the piston 42 at top dead center is defined as a zone A2, and the zone A2 is thus formed from a zone B2, which corresponds to the piston 42, and a zone C2, which corresponds to the swash plate holder 28.

The length of zone A1 in the axis L direction is set to be slightly longer than the length of zone A2 in the axis L direction, and this difference in length, that is, the distance between the top of the cylinder sleeve 41 and the top of the piston 42 at top dead center, corresponds to the dead volume. Since both the rotor 22 and the piston 42 are formed from an iron-based material, the difference in length in the axis L direction between zone B1 and zone B2 hardly changes between when the expander E is cold and when it is hot.

Whereas the swash plate holder 28 in zone C2 does not have any special cooling function, the output shaft 32 in zone C1 is cooled by a lubricating oil flowing through the interior thereof, and zone C1 therefore has a lower temperature than that of zone C2 (see FIG. 18). Moreover, whereas the output shaft 32, which is made of an iron-based material, has a small coefficient of thermal expansion, if the swash plate holder 28 were formed from an aluminum-based material having a large coefficient of thermal expansion, because of a synergistic effect thereof the thermal elongation of zone C2 when the expander E is hot would be considerably larger than the thermal elongation of zone C1. As a result, the thermal elongation of zone A2 would be larger than that of zone A1, the dead volume between the top of the cylinder sleeve 41 and the top of the piston 42 would decrease, and the volume ratio of the expander E would deviate from a set value, thus causing a degradation in the thermal efficiency.

However, in the third embodiment, since the swash plate holder 28 is formed from an iron-based material having a small coefficient of thermal expansion, the difference in thermal elongation between zone C1 and zone C2 can be decreased and, as shown in FIG. 19, a reduction in the dead volume (dead stroke) between the top of the cylinder sleeve 41 and the top of the piston 42 at top dead center can be reduced and deviation in the volume ratio of the expander E from a set value can be minimized, thus preventing the thermal efficiency from being degraded.

Moreover, since the bearing holder 99 and the swash plate holder 28 are formed from the same member, there is a contribution to a reduction in the number of components.

Although embodiments of the present invention are explained above, the present invention can be modified in a variety of ways without departing from the spirit and scope thereof.

For example, in the embodiments the expander E of a Rankine cycle system is illustrated as an example, but the rotary fluid machine of the present invention may be used in any other application and is not limited to the expander E.

In the second embodiment the casing 11 is made of an aluminum-based material, and the rotor 22, the output shaft 32, the bearing holder 99, and the swash plate holder 28 (third embodiment) are made of an iron-based material, but as long as the relationships in the size of the coefficients of thermal expansion defined in Claim 3 are satisfied, any materials other than the above-mentioned materials may be selected.

Furthermore, in the third embodiment the bearing holder 99 and the swash plate holder 28 are formed from the same member, but they may be formed from separate members.